EFFICIENCY PREDICTION ON A 2.5 MW WIND TURBINE GEARBOX

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ABSTRACT
This paper is a case study of the efficiency of a 2.5 MW wind turbine gearbox including the influence of each gearbox element: gear tooth geometry, rolling bearings and oil formulation. The power loss model used to predict the gearbox efficiency was previously validated with experimental results. The calculations showed that the efficiency of a wind turbine gearbox can be improved by selecting different wind turbine gear oil formulations, modifying gear tooth geometry. The energy savings can be even more significant if both gear tooth geometry and oil formulation are adequately selected.
KEY WORDS: wind turbine gearbox, power loss, efficiency, gear oil.

1.- INTRODUCTION
The wind energy is becoming more and more important on the energetic sustainability of the modern civilizations. Wind turbines have a significant contribution to the electrical power generation from renewal sources around the world [1]. The blades of a wind turbine rotate at very low speeds, typically 20 revolutions per minute, which are not suitable for conventional power generation using an electrical generator. This constraint is solved using a multiplying gearbox between the blades shaft and the electrical generator.
Wind turbine gearboxes handle with several megawatt and even a small efficiency increase can save energy, useful for several more households. Nevertheless, any efficiency increase will have a significant impact, reducing the power loss and the operating temperature.
The previous works [2, 3, 4, 5, 6, 7, 8, 9, 10, 11] aimed to fully characterize wind turbine gear oils in terms of physical properties and friction, on gears and rolling bearings. Experimental tests were performed, allowing to calibrate each power loss source and then a gearbox power loss model was developed.
Furthermore, the experimental results clearly showed that it is possible to increase gearbox efficiency through an improved gear tooth design or selecting the most suitable gear oil formulation, or even, combining these two possibilities.
The present work is devoted to the power loss prediction of a 2.5 MW wind turbine gearbox lubricated with different fully formulated ISO VG 320 wind turbine gear oils.

2.- WIND TURBINE GEAR OILS
Four different wind turbine gear oils were selected, covering a good range of base oils: a mineral, a polyalkylene glycol, a polyalphaolefin and a mineral+PAMA.
The physical properties of the oils can be found in Table 1. All the oils are ISO VG 320 as can be observed by the kinematic viscosity measured at 40 °C. At the usual

operating temperature of a wind turbine gear oil, 80 °C, the kinematic viscosity is quite different between oil formulations. PAGD presented more 44% viscosity than MINR and more 23% than PAOR.
A chemical characterization of each wind turbine gear oil was previously presented in [12] where the film thickness as well as the coefficient of friction were measured using a ball-on-disc device.

Table 1 – Wind turbine gear oils physical properties.

Parameter	Unit	MINR	PAOR	PAGD
Base Oil	[-]	Mineral	PAO	PAG
Density @ 15 °C	[g/cm^3]	0.902	0.859	1.059
Viscosity @ 40 °C	[cSt]	319.2	313.5	290.3
Viscosity @ 80 °C	[cSt]	43.9	60.4	78.9
Viscosity @ 100 °C	[cSt]	22.3	33.3	51.1
m_A	[-]	9.066	7.351	5.759
n_A	[-]	3.473	2.787	2.151
VI	[-]	85	150	230

3.- CALIBRATED POWER LOSS MODEL

According to Höhn *et al.* [13, 14, 15] the gearbox power loss is due to gears (no-load (P_{VZ0}) and load dependent (P_{VZP}), rolling bearings (P_{VL}), seals (P_{VD}) and auxiliary losses (P_{VX}). The auxiliary losses are considered in the case of rotating parts other than gears, bearings or seals. The total power loss is then given by equation (1).

$$P_V = P_{VZ0} + P_{VZP} + P_{VL} + P_{VD} + P_{VX} \tag{1}$$

3.1.- Rolling bearings

Rolling bearing tests were performed aiming to characterize different wind turbine gear oil formulations and rolling bearing geometries. The results were published in **[2, 3, 4, 5]**.
The rolling bearing losses were calculated using the new SKF model presented in equation (2) [16].

$$M_t = M_{rr} + M_{sl} + M_{drag} + M_{seal} \tag{2}$$

The sliding torque is dependent on the oil formulation and is estimated with equation (3).

$$M_{sl} = G_{sl} \cdot \mu_{sl} \tag{3}$$

The sliding coefficient of friction is calculated with equation (4), but the reference values of μ_{bl} and μ_{EHD} should be known in advance in order to have an accurate model.

$$\mu_{sl} = \emptyset_{bl} \cdot \mu_{bl} + (1 - \emptyset_{bl}) \cdot \mu_{EHD} \tag{4}$$

The lubrication regime influence on the coefficient of friction is calculated with the quantity presented in equation (5).

$$\emptyset_{bl} = \frac{1}{e^{2.6\times10^{-8}(n\cdot\nu)^{1.4}\cdot d_m}} \tag{5}$$

In a previous work [17] the reference values of coefficient of friction μ_{bl} and μ_{EHD} for ball and roller bearings were determined for each wind turbine gear oil. The values are presented in Table 2.

Table 2 – SKF model reference values for each wind turbine gear oil and rolling bearing type **[17]**.

Oil	Reference CoF	TBB 51107	RTB 81107
MINR	$\boldsymbol{\mu_{bl}}$	0.058	0.035
	$\boldsymbol{\mu_{EHD}}$	0.056	0.018
PAOR	$\boldsymbol{\mu_{bl}}$	0.049	0.039
	$\boldsymbol{\mu_{EHD}}$	0.044	0.010
PAGD	$\boldsymbol{\mu_{bl}}$	0.054	0.025
	$\boldsymbol{\mu_{EHD}}$	0.044	0.010

3.2.- Gears

FZG gear tests were performed using several different gear geometries lubricated with different wind turbine gear oils and the results were published in [6, 18].
The meshing gears losses were determined using Ohlendorf's equation (6) [19]. The equation takes into account the input power (P_{IN}), the local gear loss factor (H_{VL}) and the average coefficient of friction along the path of contact (μ_{mZ}).

$$P_{VZP} = P_{IN} \cdot H_{VL} \cdot \mu_{mZ} \tag{6}$$

The local gear loss factor was proposed by Wimmer [20] and is presented in equation (7).

$$H_{VL} = \frac{1}{p_b} \int_0^b \int_A^E \frac{F_N(x,y)}{F_{bt}} \cdot \frac{v_g(x,y)}{v_{tb}} \, dxdy \tag{7}$$

The coefficient of friction on the meshing gears was calculated using the Schlenck equation (8) [21].

$$\mu_{mZ} = 0.048 \cdot \left(\frac{F_{bt}/b}{v_{\Sigma C} \cdot \rho_{redC}} \right)^{0.2} \cdot \eta^{-0.05} \cdot {R_a}^{0.25} \cdot X_L \tag{8}$$

In a previous work [8], the lubricant parameter (X_L) from Schlenck equation (4), was determined based on FZG tests for each wind turbine gear oil formulation using type C gears with 40 mm of face width. The values are presented in Table 3.

Table 3 – Lubricant parameter (X_L) for each wind turbine gear oil formulation **[18]**.

Oil	MINR	PAOR	PAGD
$\boldsymbol{X_L}$	0.85	0.67	0.59

3.3.- Seals

The Freudenberg [22] equation (6) was used to calculate the seals power loss.

$$P_{VD} = 7.69 \times 10^{-6} \cdot {d_{sh}}^2 \cdot n \tag{9}$$

4.- 2.5 MW WIND TURBINE GEARBOX

A wind turbine gearbox, rating 2.5 MW was selected [23, 24]. The gearbox has a common wind turbine gearbox configuration, i.e. two planetary gear stages plus a parallel helical gear stage as presented in Figure 1. On each stage the gearbox has the rolling bearings listed on Table 4. The gear tooth geometry for each stage can be found in Table 5.

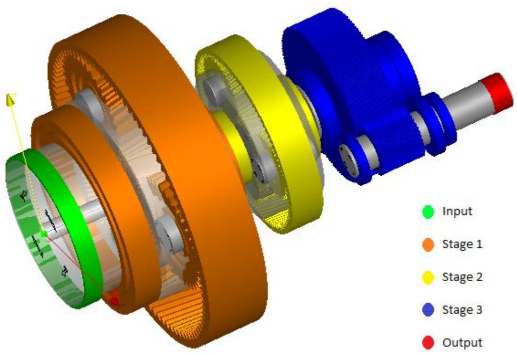

Figure 1. 2.5 MW wind turbine gearbox with two planetary gear stages plus one parallel helical gear stage.

Table 4. Rolling bearings type of the wind turbine gearbox.

Stage	**Rolling bearing type**	**Location**	**Quantity**
Stage 1	SKF NU 20/800 ECMA	Carrier	1
	SKF NU 1080 MA	Carrier	1
	SKF NU 2340 ECMA	Planets	3
	SKF NU 2340 ECMA	Planets	3
Stage 2	SKF NU 244 ECMA	Carrier	1
	SKF NU 1060 MA	Carrier	1
	SKF NNCF 4930 CV	Planets	3
	SKF NNCF 4930 CV	Planets	3
Stage 3	SKF NU 1060 MA	Pinion shaft	1
	SKF 32960	Pinion shaft	1
	SKF 32960	Pinion shaft	1
	SKF NU 1036 ML	Pinion shaft	1
	SKF NUP 236 ECMA	Wheel shaft	1
	NSK QJ 1036	Wheel shaft	1

Table 5. Gear tooth geometry of the wind turbine gearbox.

	Stage 1			**Stage 2**			**Stage 3**	
Parameter	Sun	Planet	Ring	Sun	Planet	Ring	Pinion	Wheel
z	21	35	-96	23	38	-103	117	35
b	320	320	331.5	168.4	168.4	177.4	245	240
m	16			9			7	
α_Z	20							
β_Z	10							
x	0.71	0.8031	0.2093	0.6464	0.7693	-0.0639	0.769	0.7176
S_F	1.68	1.19	1.89	1.98	1.39	2.18	2.74	2.91
S_H	1.09	1.15	1.79	1.18	1.22	2.25	2.02	1.99

4.- OPERATING CONDITIONS

The test conditions considered for the present study are resumed in Table 6. It was assumed the full power capacity of the wind turbine, i.e. 2.5 MW corresponding to an input speed on the blades of 20 rpm. The rotational and tangential speed of each gear mesh are presented in Table 7. The load conditions produced by a 1200 kNm torque applied to the input shaft, produced the maximum Hertz pressures presented in Table 7. It is important to advise that in previous works [6, 8], the operating conditions used to test fully formulated gear oils in a FZG gear testing machine were very similar to those presented here.

Table 6. Operating conditions considered for the wind turbine gearbox.

Condition	**Quantity**
Input torque	1.2 MNm
Input speed	20 rpm
Output speed	2040 rpm
Nominal power	2.5 MW
Operating temperature	60 and 80 ˚C
Lubrication method (gears)	Oil jet
Lubrication method (bearings)	Dip

Table 7. Rotational and tangential speed on the gear mesh of a wind turbine gearbox for an input speed of 20 rpm.

		Stage 1		Stage 2		Stage 3
Property	Unit	P/S	P/R	P/S	P/R	Helical
n	[rpm]	111.4	34.9	610.4	190.6	610.4
v_t	[m/s]	1.867	0.974	6.302	3.251	24.933
p_0	[MPa]	1028	699	921	624	567
(P/S – Planet/Sun, P/R – Planet/Ring)						

4.- SIMULATION PROCEDURE

To carry on the simulations, the power loss model was used as resumed in equation (10).

$$P_V = P_{VZP} + P_{VL} \tag{10}$$

The no-load losses of gears and seals were disregarded for different reasons. In the present study, the no-load gear losses will not be considered in the simulation since the models available are not independent of the gearbox configuration. Furthermore, the experimental and model results presented in previous works [6, 18] show that the influence of the no-load gear losses on the total torque loss of a gearbox, at low speed, are very small. At the same time, the oils used are ISO VG 320 and the differences between them, in terms of no-load losses, are expected to be very small.

The seal losses were not considered since the seals used in this particular gearbox are not known. Furthermore, the Simrit equation (9) does not account for the influence of different oil formulations.

A simulation was performed for MINR, PAOR and PAGD gear oils. Two different operating temperatures were considered, 60 and 80 ˚C which is the usual range of operation in a wind turbine gearbox.
The first and second stage were analysed using the concept of mesh-power, while stage 3 of the wind turbine gearbox, being a parallel helical gear, was analysed using equation (6).
The input power on each planetary stage is splitted in 3 planets and the tangential force applied on the base plane is calculated with equation (11).

$$F_{bt} = \frac{P_{IN}}{3 \cdot v_t} \tag{11}$$

The mesh power in each meshing pair should be calculated as presented in equation (12), and so the relative speed was considered. The mesh power (P_M) should be used in equation (6) instead of input power (P_{IN}) for the case of planetary gears. Regarding the coefficient of friction the sum velocities in the pitch point ($v_{\Sigma C}$) should also be calculated using the relative velocities.

$$P_M = F_{bt} \cdot v_t{}' \tag{12}$$

The input shaft of stage 3 runs at 610 rpm, which corresponds to 25 m/s of tangential speed.
Independently of the oil used, the gears will perform under full-film conditions. The Schlenck equation is suitable for mixed film lubrication conditions and the coefficient of friction would decrease ad infinitum if the speed is increased without care. To avoid the underestimation of the meshing gears power loss, the third stage coefficient of friction was calculated for Λ=2, i.e. it was assumed that the coefficient of friction is better estimated if calculated at the speed corresponding to the beginning of full film conditions.
The rolling bearing power losses were calculated using the calibrated power loss model described before. The coefficients of friction (μ_{bl} and μ_{EHD}) were determined based on the experimental results are here again used for the simulation performed, assuming that no significant difference is found between 60 ˚C and 80 ˚C [17].

5.- SIMULATION RESULTS

Considering the main sources of power loss in each gearbox stage, gears and rolling bearings, antagonistic effects were observed, as presented in Figure 2, Figure 3 and Figure 4. PAGD reduced the gears power loss but slightly increased the rolling bearing losses. The opposite behaviour is observed for MINR.
The temperature also has an opposite effect, depending if gears or bearings are considered. Increasing the operating temperature increases the gear losses, as shown in Figure 3a) and Figure 3b). The rolling bearing losses reduce by increasing the temperature and consequently lowering the viscosity. The rolling torque (M_{rr}), in rolling bearings, is the main source of power loss in stage 3 and it is mainly dependent on speed and viscosity. Consequently, the rolling bearings power loss in stage 3 is almost independent of the oil formulation.
The present results showed that ism possible to reduce the total power loss by changing the oil formulation used as presented in Figure 2.

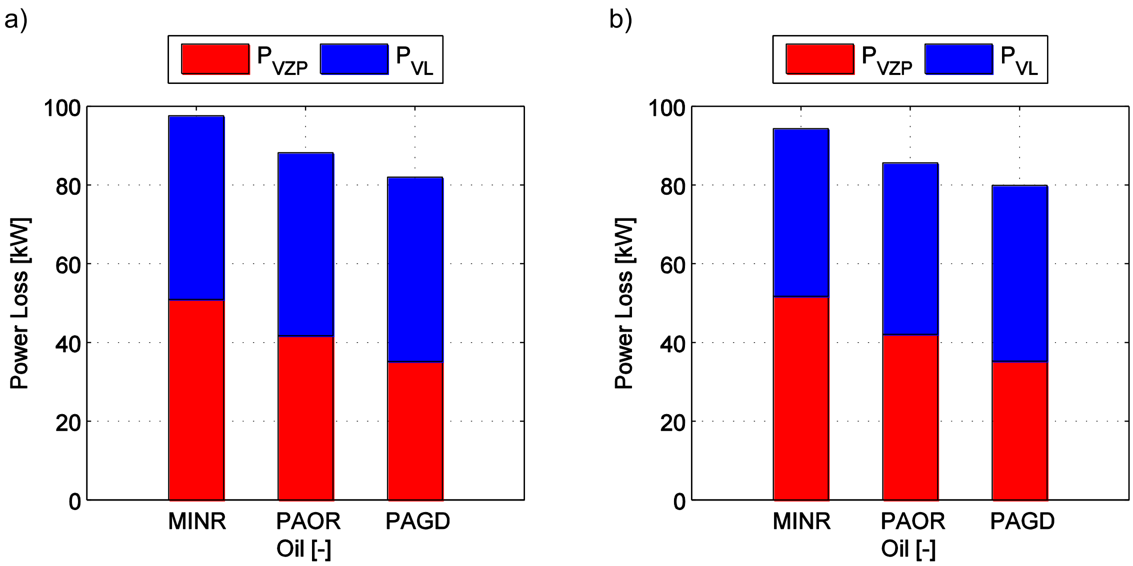

Figure 2. Total power loss results: a) 60 ˚C and b) 80 ˚C **[12]**.

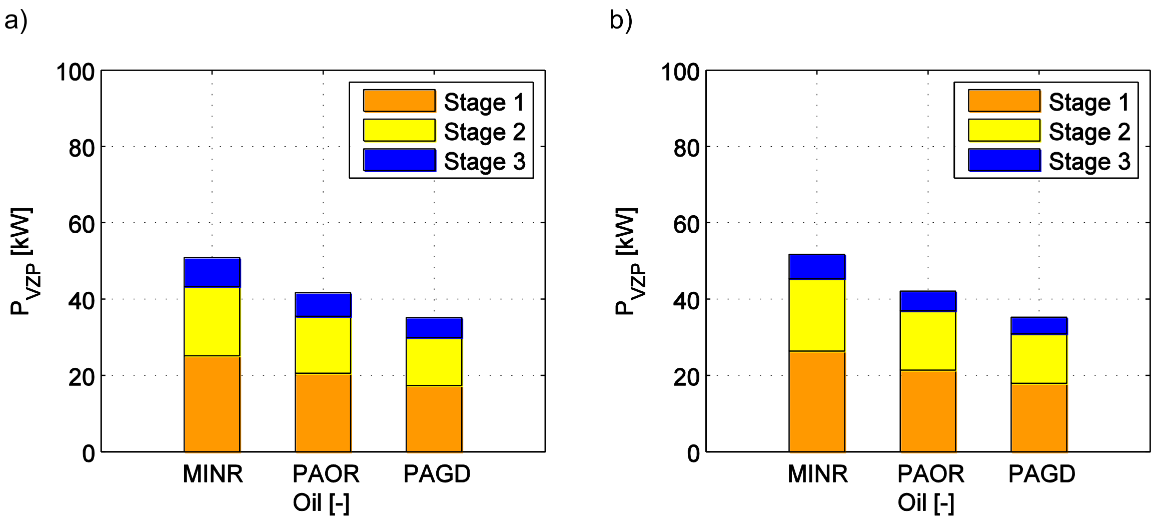

Figure 3. Meshing gears power loss results: a) 60 ˚C and b) 80 ˚C **[12]**.

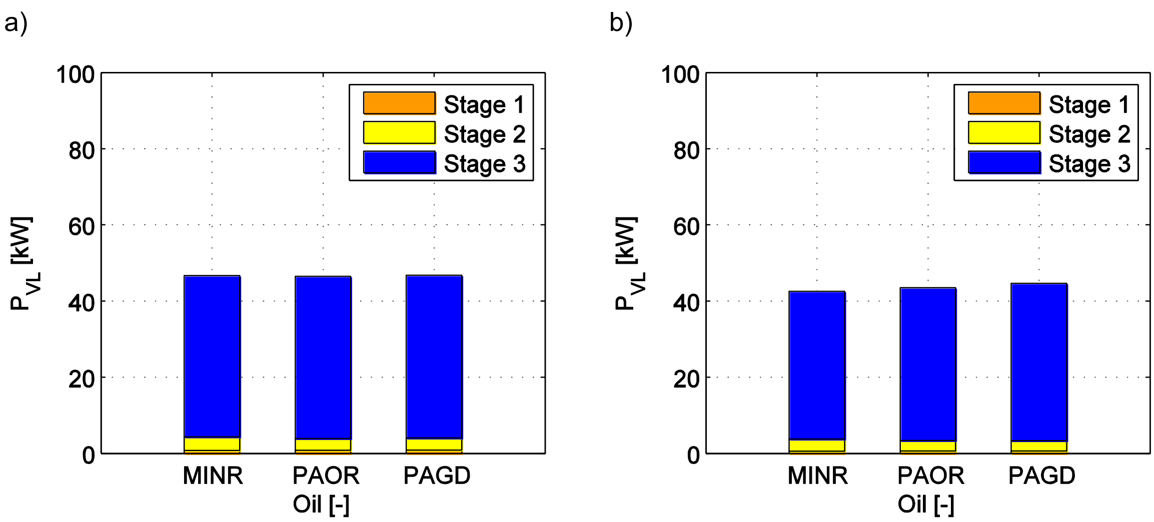

Figure 4. Rolling bearings power loss results: a) 60 ˚C and b) 80 ˚C **[12]**.

The efficiency of each gearbox stage is presented in Table 8 for each oil formulation.

Table 8. Wind turbine gearbox efficiency [%] and total power loss for each oil formulation at 80 ˚C.

Oil	Gearbox design	Stage 1	Stage 2	Stage 3	Global	P_V[W]
MINR	Standard	98.93	99.12	98.20	96.25	93,597
PAOR	Standard	99.12	99.28	98.20	96.59	85,043
PAGD	Standard	99.26	99.38	98.18	96.82	79,423

6.- CONCLUSIONS

The power loss model proved to be a valuable tool to optimize the gearbox efficiency. It was found that the rolling bearing losses predominate in very high speed conditions while meshing gear power losses are very important in low and intermediate speeds of stage 1 and 2 planetary sets. The results showed that a PAGD can promote an efficiency increase up to 0.6% when compared with a MINR.

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NOTATION AND UNITS

a_A ASTM D341 reference kinematic viscosity [cSt]
a' centre distance [mm]
b gear face width [mm]
d_m rolling bearing mean diameter [mm]
F_N gear normal force per unit contact length in each meshing position along the path of contact [N/mm]
F_{bt} gear tangential force on the base plane [N]
G_{rr} rolling torque factor depending on the bearing type, bearing mean diameter and applied load [/]
G_{sl} sliding torque factor depending on the bearing type, bearing mean diameter and applied load [/]
LSS low speed shaft [/]
LIS low intermediate shaft [/]
HSS high speed shaft [/]
H_{VL} local gear loss factor [/]
i gear ratio [/]
m gear module [mm]
m_A ASTM D341 viscosity parameter [/]
M_{rr} rolling friction torque [Nmm]
M_{sl} sliding friction torque [Nmm]
M_{drag} friction torque of drag losses [Nmm]
M_{seal} friction torque of seals [Nmm]

M_t internal bearing friction torque [Nmm]
n_A ASTM D341 viscosity parameter [/]
n rotational speed [rpm]
P_{IN} input power [W]
P_V total power loss [W]
P_{VZ0} no-load gears power loss [W]
P_{VZP} meshing gears power loss [W]
P_{VL} rolling bearings power loss [W]
P_{VD} seals power loss [W]
p_b gear transverse pitch [mm]
R_1 geometry constant for rolling friction torque [/]
R_a average surface roughness [m]
S_1 geometry constant for sliding friction torque [/]
S_F root stress safety factor [/]
S_H flank stress safety factor [/]
v_g gear sliding velocity in each meshing position along the path of contact [m/s]
v_{tb} gear tangential velocity on the base plane [m/s]
$v_{\Sigma C}$ sum of the gear surface velocities on the pitch point [m/s]
x_z gear profile shift [/]
z gear number of teeth [/]
α_z gear pressure angle [˚]
β_z gear helix angle [˚]
$\emptyset_{bl}$ sliding friction torque weighting factor [/]
$\emptyset_{ish}$ inlet shear heating reduction factor [/]
$\emptyset_{rs}$ kinematic replenishment/starvation reduction factor [/]
Λ specific film thickness [/]
μ_{bl} coefficient of friction in boundary film lubrication [/]
μ_{EHD} coefficient of friction in full film lubrication [/]
μ_{sl} sliding coefficient of friction [-]
υ kinematic viscosity [cSt]

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